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1987/04

TR-TM-004
NRC NO. 27948

IMPACT AND FRETTING WEAR STUDIES
PART 1 — TEST RIG AND REVIEW OF THEORIES

P.L. Ko

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IMPACT AND FRETTING WEAR STUDIES

Part 1 - TEST RIG AND REVIEW OF THEORIES

ÉTUDES DE L'USURE PAR IMPACT ET DE L'USURE DE CONTACT

Partie 1 - BANC D'ESSAIS ET ÉTUDE DES THEORIES

P.L. Ko

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TR-TM-004
NRC NO.27948

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ABSTRACT

A wear test rig, capable of studying reciprocating, sliding and impacting wear at temperatures up to 300°C, has been designed and constructed in NRC's Tribology and Mechanics Laboratory. Previous studies have gained information on the most serious combinations of sliding /impacting situations. This new test rig will enable the reciprocating and impacting conditions to be controlled more independently so that a wider range of precisely controlled operating conditions can be simulated in the laboratory. The wear and damage data thus obtained, together with the recorded information on the dynamic parameters, will help the formulation of more accurate predictive models for better assessments of the useful life of impact-sliding components in practical situations.

RÉSUMÉ

On a conçu et construit, au Laboratoire de tribologie et de mécanique du CNRC, un banc d'essais permettant d'étudier le glissement alternatif et l'usure par impact à des températures atteignant 300°C. Les études précédentes ont permis de recueillir des informations sur les plus importantes combinaisons de conditions de glissement/impact. Le nouveau banc d'essais permettra de contrôler, de façon plus indépendante, les conditions de mouvement alternatif et d'impact, de sorte qu'une plus vaste gamme de conditions d'utilisation contrôlées avec précision pourront être simulées dans le laboratoire. Les données relatives à l'usure et aux dommages ainsi obtenues, combinées aux données recueillies sur les paramètres dynamiques, permettront de formuler des modèles de prévision plus précis en vue de mieux évaluer la durée utile des éléments soumis à des conditions d'impact-glissement dans des situations concrètes.

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LIST OF SYMBOLS

SYMBOL		Units
a	maximum radius of contact area	m
r	intantaneous radius of contact area	m
$f(r)$	tangential traction	N
E, E_i	Modulas of Elasticity	$N.m^{-2}$
F	normal impact force	N
F_μ	tangential force during gross slip, = μF	N
F_T	tangential force due to elastic deflection,	N
G	modulus of rigidity	$N.m^{-2}$
I	moment of inertia of the sphere about its centre	m^4
I_p	impulse	N.s
I_E	energy or work	N.m
K	non-dimensional radius of gyration	
k	stiffness	$N.m^{-1}$
m, m_i	mass of the sphere	kg
$p(r)$	normal contact pressure	Pa
p_o	maximum normal pressure	Pa
R	radius of the sphere	m
R'	equivalent radius of two spheres	m
s	shear stress	$N.m^{-2}$
t	time	s
u_x, u_y	tangential velocities	$m.s^{-1}$
U_x, U_y	tangential displacements of the surfaces	m
U	linear speed of moving plate	$m.s^{-1}$
v_z	velocity in the normal direction, i.e., z	$m.s^{-1}$
v_i	initial value of v_z	$m.s^{-1}$
V	oblique impact velocity	$m.s^{-1}$
X,Z	maximum elastic deflections	m
α,α_z	relative normal approach	m
α_m	maximum normal approach	m
β	surface damage contribution factor	
y	ratio of ω_x/ω_z	
σ_y	uniaxial tensile yield stress	$N.m^{-2}$
μ	coefficient of friction	
ν_1,ν_2	Poisson's ratio	
ω	wear factor	
ω_x, ω_z	natural frequencies in the x,z directions	Hz
ϕ	angle of incidence	deg
τ	contact time	s

IMPACT AND FRETTING WEAR STUDIES

Part 1 - TEST RIG AND REVIEW OF THEORIES

1.0 INTRODUCTION

In many mechanical components some form of vibration is unavoidable. Fretting wear may occur in components where the contacting parts are held together either by a normal load or with a tight tolerance fit. In situations where a clearance exists, such as that between the tube and its support in tube-and-shell heat exchangers, or between loosely supported parts, wear may be a result of impacting and rubbing.

Examples of fretting wear, impact wear, or wear due to a combination of impacting and reciprocating rubbing are numerous. They can be found in small components, such as types and ribbons in typing machines, and electric contacts; as well as in large components, such as the sucker rod in oil wells and swivel connections in offshore equipment. Other examples can be found in mining and forging operations; in the wear of wire ropes, gear teeth, claws on pinions and couplings, etc. Impact and fretting wear is also a common tribology problem in a wide range of power plant machinery, such as in steam generators and heat exchangers, reactor fuel rods and control rods.

While some of these wear problems have fundamental significance, many others have economic and safety consequences as well. In nuclear power plants, for example, failure of steam generator tubes can result in costly repairs and safety concerns.

To understand the mechanics and the various mechanisms involved a test rig has been designed for the studies of impact, fretting and reciprocating sliding wear. This report will describe the new test rig and will review the theoretical background on impact and fretting wear. The overall research program, which includes both theoretical and experimental studies, will also be described.

2.0 LITERATURE SURVEY

2.1 Fretting - Reciprocating Sliding

Fretting arises when contacting surfaces undergo oscillatory tangential displacement of small amplitude. According to Waterhouse (1972) it is concerned with amplitudes of less than 25 μm and certainly not greater than 130 μm. It represents the interaction of several mechanisms. In a review, Hurricks(1970) summarized fretting in three stages. It is initiated by adhesion and subsequently, when the wear particles are oxidized, abrasion becomes the primary mechanism. The cycle is repeated after the dispersion of the wear particles. Therefore,

the ease with which wear particles are dispersed has a significant effect on the rate of fretting wear. Waterhouse(1977) suggested that delamination could be responsible for the final generation of fretting particles.

The delamination wear theory was introduced by Suh(1973). It is based on the behaviour of dislocations at the surface, subsurface crack and void formation, and on the subsequent joining of cracks by shear deformation of the surface. Accordingly, the tangential traction is a very important factor, as it induces tensile stresses below the surface and thus contributes strongly to crack nucleation and growth.

Based on the observed mode of metallic material removal, Sproles et al(1978) also believed that delamination rather than abrasion, welding, and metal transfer is the predominant mechanism for fretting wear.

Johnson(1963) studied the mechanics of microslip as the fundamental cause of fretting. The oscillating tangential force is a controlling parameter. He suggested that the wear rate of some component due to fretting, such as lap-joints, may follow a cube law as a function of amplitude similar to that between the energy loss per cycle and the vibration amplitude. The latter was based on the work of Goodman(1959) which shows that the slip region increases in size roughly in direct proportion to the amplitude of vibration.

Stowers and Rabinowicz(1973) identified two different regimes of fretting as a function of amplitude. In the low amplitude regime the volumetric wear per cycle is a function of the amplitude squared, whereas, in the high amplitude regime, it is directly proportional to the amplitude. They suggested that material loss resembles more closely that produced by uni-directional adhesive wear than that produced by other modes of wear, and the amount of wear can be computed by Archard's equation.

2.2 Impact Wear

Engel(1978) classified impact wear into two types, namely: percussion and erosion. Percussive impact wear, which is often simply referred to as impact wear, is caused by repeated solid body impacts. Erosive impact wear results from the impact of streams of jets of solid particles or liquids on a surface. The most common mechanism of impact wear is by the formation of subsurface and/or surface cracks (Engel,1976). Erosive wear, on the other hand, is believed to occur by a combination of abrasion and deformation.

The term 'compound impact' is used to define a condition of impact and sliding. Under this mode, the shear stress becomes an important factor affecting the wear damage (Ko,1985; Rice,1979). Engel(1976) and his colleagues considered the wear energy to be a fraction of the peak strain energy of the impact cycle and thus purely stress-dependent. The results of their experiments using a projectile showed that pure normal impacting slowly produces an initial wear phenomenon; the larger the sliding speed component, the sooner wear is initiated. The contribution of the normal component is small compared with that of the sliding

component. In the absence of a sliding speed component, the stress state at the projectile apex is nearly hydrostatic, allowing only small shear stress near the surface. They concluded that the stress ratio, i.e., peak stress over yield stress, and the sliding speed are two crucial parameters for impact wear.

Rice, et al(1980a) established that during sliding and compound impact, subsurface material structures are altered, giving rise to wear. The altered characteristics of the subsurface zones depend on the magnitude of these impacts together with the type of materials and the environment. The impact stresses give rise to strain gradients and subsurface plastic deformation, and potentially lead to void and crack generation. They also found that the relative transverse sliding velocity is a significant parameter acting on the wear of titanium alloys under repetitive compound impact (Rice, et al,1980b), and that wear increases as the peak normal impulsive stress is increased.

In a study of impact wear of ductile metals, Rabinowicz and Hozaki(1985) found that wear due to oblique impacting at an angle of 24° from normal was increased by a factor of 9. They concluded that impact wear is not itself a form of wear but merely describes one way in which the normal force and shearing action required to produce friction and wear can be applied. Depending on the circumstances, any type of wear could account for the major proportion of the material removed. Since adhesive wear is in ordinary sliding the most important form of wear, it is, therefore, also very prominent during impact conditions. A similar conclusion was arrived at by Ko et al(1982) in their studies of heat exchanger tube fretting wear. Based on scanning electron photomicrographs of worn surfaces, and on impact force analyses they found that the primary cause of wear in all types of tube motion is due to shear.

2.3 Predictive Models

Goldsmith(1960) provided a very thorough treatment of the impact theories. The essential impact parameters are deduced from the energy and momentum laws, and from the relations for the dynamic contact of two bodies in the elastic and plastic regions. Based on the Hertz theory, the forces and elastic deformations in the contact point are determined. In the plastic case, a simple relation between force and the depth of penetration exists.

Several predictive models of the compound impact wear process for two constrained bodies were developed by combining Archard's wear equation with equations derived from the conservation of energy and momentum.

Fricker(1972) calculated fretting wear rates of steam generator tubes by first, deriving an equation for energy input per cycle from excitation force and the vibration amplitude (mid-span displacement). Then, by assuming the wear energy equals the energy due to excitation, he equated the excitation energy to the Archard's wear equation.

Blevins' predictive model(Blevins,1983) for tube wear was based on Engel's impact wear theory. In Blevins' model, Hertzian theory for a point contact was used to model the stress field and the compli-

ance between impacting surfaces. The dynamic contact force and stress were obtained by equating the potential energy due to deformation with the kinetic energy from the impact velocity. The contact stress was expressed as a function of mid-span peak-to-peak amplitude, frequency, tube diameter, modulus of elasticity and the equivalent tube mass.

Ko(1984,1985) correlated the impact-fretting wear rate with a function that consists of the shear component of the impact force and the accumulated contact time. A somewhat similar model by Frick, Sobek and Reavis(1984) modified the Archard's equation by replacing the normal force and sliding distance with a work rate (work per unit time) term and the total wear process time.

A very thorough treatment of the force-sliding distance model for impact wear was given by Levy and Morri (1985). In essence, impact wear was treated as an elastic-contact dominated process, with load and relative slip during contact as the most crucial governing parameters. The load-sliding distance integral was considered to be directly dependent on the incident kinetic energy.

Wellinger and Breckel(1969) investigated the correlation of impact energy and metal deformation. The parameters included impact force, duration, impact coefficient, deformation energy and depth of penetration. They found that, for soft materials, the wear and the energy that was mainly used for the flow of the material follow a power law as a function of the impact velocity.

3.0 THEORETICAL BACKGROUND

Engel(1974) briefly discussed three general categories of impact analysis. These are: 1. stereomechanical analysis, 2. Hertz (quasi-static) analysis, and 3. analysis including vibrational effects.

Rigid bodies are considered in the first category. The total impulse

$$I = \int F(t)\, dt$$

and departing velocities can be calculated from the initial momenta. The coefficient of restitution, e, may be used to account for the modified rebound speeds. This method does not yield information about the magnitude of force during impact nor the impact time.

The Hertz method (Timoshenko et al, 1969) considers local deformations and allows for the elastic approach between the impact bodies during the impact but ignores the stress wave effects in the impact bodies. This method leads to good results for the impact of spheres and, in general, for stubby bodies where the contact duration is much

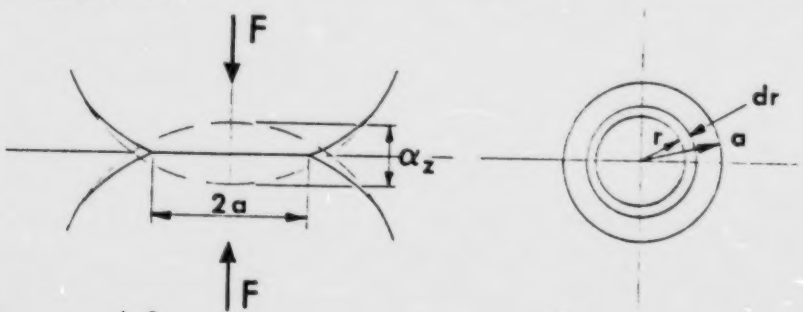
longer than the half period of the lowest natural mode of vibration, thus avoiding amplification. It yields complete impact histories.

For impact cases where basic vibrational frequencies of the bodies are low, the vibrational motion throughout the bodies must be considered and the forcing function is a time varying contact force at their junction.

In the following, oblique impact and wear theories, developed by three groups of researchers, are discussed. These are preceded by a brief outline of Hertz's theory for normal impact between elastic bodies, e.g., sphere on sphere or sphere on flat.

3.1 Impact of Elastic Spheres (Goldsmith, 1960)

For elastic impacts, the force-deformation law at some instant during the impact, is



$$F = k_2 \alpha_z^{1.5} \qquad (1)$$

where $\alpha_z = a^2/R'$ is the relative normal approach

$$k_2 = 4R'^{0.5}/3\pi(\zeta_1+\zeta_2)$$

and $$R' = R_1R_2/(R_1+R_2)$$

$$\therefore \quad F = 4a^3/3\pi R'(\zeta_1+\zeta_2) \qquad (2)$$

where $\zeta_1 = (1-\nu_1^2)/\pi E_1$

$\zeta_2 = (1-\nu_2^2)/\pi E_2$

For like spheres, $R_1 = R_2 = R$, $R' = R/2$ and $\zeta_1 = \zeta_2$

$$\therefore \quad F = 4a^3E/3R(1-\nu^2) \qquad (3)$$

The pressure distribution across the circular contact area of radius a is

$$p = p_o(a^2-r^2)^{0.5}/a \qquad (4)$$

where the maximum pressure, $p_o = 3F/2\pi a^2$

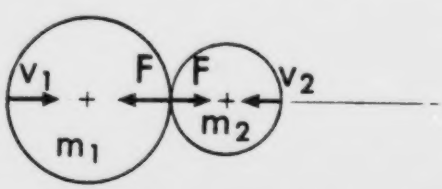
and r is the radial distance from the centre of contact surface.

$$p(r) = 2E(a^2-r^2)^{0.5}/\pi R(1-\nu^2) \qquad (5)$$

$$= 4G(a^2-r^2)^{0.5}/\pi R(1-\nu)$$

where $G = E/2(1+\nu)$

For elastic impact with a known impact velocity :



$$F = -m_1\dot{v}_1 = -m_2\dot{v}_2 = -[m_1m_2/(m_1+m_2)]\ddot{\alpha} = k_2\alpha^{1.5}$$

The relative approach velocity $\dot{\alpha} = v_1+v_2$. At the instance of maximum compression, α_m, $\dot{\alpha} = 0$. At the beginning of impact, the initial relative approach velocity is v_i, i.e., $\dot{\alpha} = v_i$. From these conditions, the following equations are derived (Goldsmith, 1960).

$$t = 4.53\ (\zeta_1+\zeta_2)^{0.4}\ m^{0.4}\ R^{-0.2}\ v_i^{-0.2} \qquad (6)$$

$$\alpha_m = 1.54\ v_i^{0.8}(\zeta_1+\zeta_2)^{0.4}\ m^{0.4}\ R^{-0.2} \qquad (7)$$

A reasonable approximation for α is given by Hunter (1957):

$$\alpha = \alpha_m \sin(1.068\ v_i t/\alpha_m) \qquad (8)$$

and $$F = 1.14\, v_i^2 \sin(1.068\, v_i t/\alpha_m)/n_1\alpha_m \qquad (9)$$

$$\text{for} \quad 0 \leq t \leq (2.942\alpha_m/v_i) ;$$

$$= 0 , \qquad \text{for} \quad t > 2.942\alpha_m/v_i$$

where $m = m_1 m_2/(m_1+m_2)$

and $n_1 = 1/m = 3/(4\pi\rho R^3)$ for a sphere on a massive plane surface

3.2 The Oblique Impact Theory (Maw et al, 1976)

For the oblique impact of a sphere of mass m on a plane plate of infinite mass of similar materials, Maw et al(1976) developed a set of equations for forces and velocities at various incident angles. They defined the conditions for gross-slip and sticking (lock-up) in terms of the coefficient of friction and the tangential velocity. The latter was derived from an iteration technique based on the general impact theory.

From Equations (5) and (4), and for impacts between a sphere and a plane plate of infinite mass of similar materials:

$$p(r) = 2G(a^2-r^2)^{0.5}/\pi R(1-\nu) \qquad 0 \leq r \leq a \qquad (10)$$

$$= 0 \qquad\qquad r > a$$

and $$F = 2a^3E/3R(1-\nu^2) = 4a^3G/3R(1-\nu) \qquad (11)$$

During gross slip the distribution of traction is :

$$f(r) = \mu p(r) = 2\mu G(a^2-r^2)^{0.5}/\pi R(1-\nu) \qquad (12)$$

and the tangential displacement at the centre of the contact circle due to this traction, following the derivation from Mindlin(1949), is :

$$U_x = \mu(2-\nu)a^2/2R(1-\nu) = \mu(2-\nu)\alpha_z/2(1-\nu) \qquad (13)$$

If the sphere is on the point of sticking,

$$u_x = dU_x/dt = \mu(2-\nu)d\alpha_z/2(1-\nu)dt$$

$$= \mu(2-\nu)v_z/2(1-\nu) \qquad (14)$$

since the initial value of v_z is v_i , impact will commence with a central stuck region if :

$$u_x < \mu(2-\nu)v_i/2(1-\nu) \qquad (15)$$

Defining a non-dimensional angle of incidence

$$\Psi = 2(1-\nu)u_x/\mu(2-\nu)v_i \qquad (16)$$

where u_x is the corresponding tangential velocity, initially,

$$\tan\phi = u_x/v_i.$$

the condition for **sticking** is, from Fquation (15)

$$\Psi_1 < 1 \qquad (17)$$

If the angle of incidence is sufficiently large, the whole impact may take place in gross slip and the tangential force is, at all times, multiplied by the normal force. The final tangential velocity is

$$u_x = u_x' - \Delta u_x$$

here Maw(1976) defined

$$\Delta u_x = u_{x_1} - u_{x_2} = 2\mu v_i(1+K^{-2})$$

thus $$u_x = [\mu(2-\nu)v_i\Psi_1/2(1-\nu)] - 2\mu v_i(1+K^{-2}) \qquad (18)$$

where $K = (I/mR^2)^{0.5}$ is a non-dimensional radius of gyration, and the final normal velocity is $-v_i$, i.e., $v_z = -v_i$.

The tangential force is established as

$$F_\mu = m\, du_x/(1+1/K^2)dt \qquad (19)$$

The condition for **gross slip** to occur throughout the cycle is

$$u_{xslip} > u_{xstick}$$

or $[\mu(2-\nu)v_i\Psi_1/2(1-\nu) - 2\mu v_i(1+K^{-2})] > \mu(2-\nu)v_z/2(1-\nu)$

i.e. $\Psi_1 > [4(1-\nu)(1+K^{-2})/(2-\nu)] - 1$ (20)

Further defining $X = (1-\nu)(1+K^{-2})/(2-\nu)$

$\Psi_1 > 4X - 1$ for gross slip (21)

The ratio of the tangential stiffness to the normal stiffness, according the Mindlin[28], is

$$k_T/k_N = (1-\nu)/(1-\nu/2) \quad (22)$$

and the ratio of the corresponding natural frequencies is

$$\omega_T/\omega_N = [(1-\nu)(1+1/K^2)/(1-\nu/2)]^{0.5} = (2X)^{0.5} \quad (23)$$

which has a value greater than 1 for solid spheres and steel disks (Maw,1976; Johnson,1985). This implies that the tangential force almost completes a full cycle during the time when the normal force goes through half a cycle.

3.3 Impact Wear Theory (Levy et al,1985)

In Levy's theory, a load-sliding distance integral is developed in terms of incident angle and impact velocity. This wear integral is used to correlate with the experimental wear volume. They also used the coefficient of friction and tangential velocity, as that used in Maw's theory, to define the conditions for lock-up. The effect of elasticity is included in setting up the conditions.

Impulse $I_p = \int F\,dt = Ft$

$$= -\int_i^r m(dv/dt)dt = m(v_i - v_r) \quad (24)$$

The integration of the impact force F over the time t gives the momentum change I_p, in the struck body.

An impact coefficient, e, may be defined as :

$$e = -(v_r/v_i) \tag{25}$$

where $e = 1$ for a pure elastic impact

and $e = 0$ for a complete plastic impact

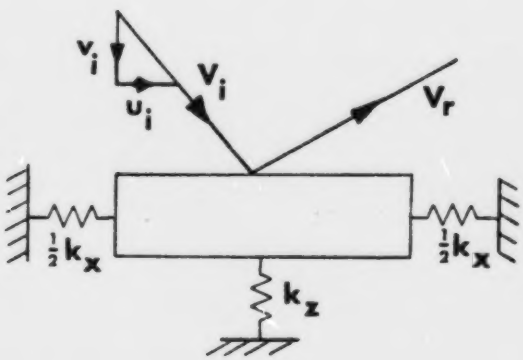
thus $v_r = -ev_i$

and $I_p = m(v_i + ev_i) = mv_i(1 + e)$

or $e = (I_p/mv_i) - 1$

Energy (work)
$$I_E = \int_0^{\alpha} F\,dz = -\int_0^{\alpha} m(dv/dt)dz = -\int_i^r m\,vdv$$
$$= m(v_i^2 - v_r^2)/2 = mv_i^2(1 - e^2)/2 \tag{26}$$

The above treatment may be extended to include the horizontal components as in a compound impact.



Impulse
$$I_{p\mu} = \int F_\mu\,dt = F_\mu\,\tau$$
$$= -\int_i^r m(du/dt)dt = m(u_i - u_r) \tag{27}$$

If we take the ratio of Equs.(24) and (27), a definition for the average coefficient of friction is obtained:

$$\mu = I_{p\mu}/I_p = F_\mu/F = (u_i - u_r)/(v_i - v_r) \tag{28}$$

The energy dissipated through friction sliding is

$$\text{Energy} \quad I_{E\mu} = \int F_\mu ds = \int_i^r m\, u\, du$$

$$= m(u_i^2 - u_r^2)/2 \tag{29}$$

If we define the load-sliding distance integral as

$$\mu I_w = \int F_\mu ds \tag{30}$$

then $$\mu I_w = m(u_i^2 - u_r^2)/2$$

or $$I_w = m(u_i^2 - u_r^2)/2\mu \tag{31}$$

Substituting Equ.(28) into Equ.(31) yields

$$I_w = m(u_i + u_r)(v_i - v_r)/2$$

$$= mv_i(1+e)[2u_i - \mu(1+e)v_i]/2$$

Since $v_i = V\sin\phi$ and $u_i = V\cos\phi$

$$I_w = mV\sin\phi(1+e)[2V\cos\phi - \mu(1+e)V\sin\phi]/2$$

$$= mV^2(1+e)[\sin 2\phi - \mu(1+e)\sin^2\phi]/2 \tag{32}$$

Defining the impact energy (incident kinetic energy), $I_e = mV^2/2$, a dimensionless wear integral J, may be written as

$$J = I_w/I_e = (1+e)[\sin 2\phi - \mu(1+e)\sin^2\phi] \tag{33}$$

3.3.1 Elasticity Effects

$$F(t) = m\omega_z^2 Z_o \sin(\omega_z t) \tag{34}$$

since $m\omega_z^2 = k_z$, and $m\omega_x^2 = k_x$

Frictional resistance $F_\mu = \mu k_z Z_o \sin(\omega_z t)$ (35)

and the tangential force due to elastic deflection in the x-dir.

$$F_T = k_x X_o \sin(\omega_x t) \quad (36)$$

For **lock-up**, i.e., **no sliding**, one of the conditions is

$$F_\mu > F_T$$

$$\text{i.e.,} \quad \mu\omega_z v_i \sin(\omega_z t) \geq \omega_x u_i \sin(\omega_x t) \quad (37)$$

for $v_i = Z_o\omega_z$ or $Z_o = v_i/\omega_z$ and $k_z = m\omega_z^2$

$u_i = X_o\omega_x$ or $X_o = u_i/\omega_x$ and $k_x = m\omega_x^2$

also $v_i/u_i = \tan\phi$

Eq(37) may be written as

$$\mu\tan\phi \geq \omega_x \sin(\omega_x t)/[\omega_z \sin(\omega_z t)] \quad (38)$$

Defining $y = (k_x/k_z)^{0.5} = \omega_x/\omega_z$, Eq(38) becomes

$$\mu\tan\phi \geq y\sin(y\omega_z t)/\sin(\omega_z t) \quad (39)$$

for $t \ll 1$, $\sin(y\omega_z t)/\sin(\omega_z t) = y$

$\therefore \quad \mu\tan\phi \geq y^2$

the above equation can also be arrived at by combining Equations (17) and (22).

The lock-up condition is valid for the fraction of the impact period when the two deflection force curves, F_μ and F_T, intersect. From there on the two surfaces are subject to gross sliding throughout the remaining fraction of the impact period.

The condition for this change in the surface interaction behaviour is predicted by inserting an equality in Eq(39) and solving

for time $t = t_s$. Substituting $\tau_s = \omega_z t_s$ (time from lock-up to slide) gives

$$\mu\tan\phi = \gamma\sin(\gamma\tau_s)/\sin\tau_s \tag{40}$$

During lock-up, the tangential velocity u_i may be obtained by calculating the retardation due to the linear spring k_x.

$$m(du/dt) = -k_x X_o \sin(\omega_x t)$$

$$u_s - u_i = -\omega_x^2 X_o \int_0^{t_s} \sin(\omega_x t)dt$$

$$= \omega_x X_o[\cos(\omega_x t_s) - 1] = u_i[\cos(\gamma\tau_s) - 1]$$

$$u_s = u_i\cos(\gamma\tau_s) \tag{41}$$

During gross sliding

$$m\int_s^r (du/dt)dt = -\mu k_z Z_o \int_s^r \sin(\omega_z t)dt$$

$$u_r - u_s = \mu\omega_z Z_o[\cos(\omega_z t_r) - \cos(\omega_z t_s)]$$

$$= \mu v_i[-1 - \cos\tau_s]$$

$$\therefore\ u_r = u_i\cos(\gamma\tau_s) - \mu v_i(1+\cos\tau_s) \tag{42}$$

Taking into account explicitly the coefficient of restitution e, Eq(42) can be written as

$$u_r = u_i\cos(\gamma\tau_s) - \mu u_i\tan\phi(e+\cos\tau_s)$$

$$= u_i[\cos(\gamma\tau_s) - \mu\tan\phi(e+\cos\tau_s)] \tag{43}$$

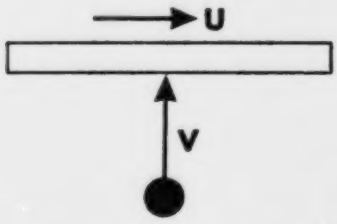
Substituting Eq.(31) into Eq.(43) and replacing u_i by $V\cos\phi$, we have

$$I_w = mV^2\cos^2\phi \{1 - [\cos\gamma\tau_s - \mu\tan\phi(e+\cos\tau_s)]^2\}/2\mu$$

or $$J = \cos^2\phi\{1 - [\cos\gamma\tau_s - \mu\tan\phi(e+\cos\tau_s)]^2\}/\mu \quad (44)$$

For the limiting case of $\tau_s \to 0$, Eq.(44) reduces to the form in Eq.(33), i.e., with no lock-up.

3.4 Engel's Impact Wear Model (Engel,1973)



A mass impacts with a normal speed V on a plate moving at a linear speed U. Upon hitting the plate, the impacting mass m is accelerated with a horizontal component due to the friction force. The instantaneous speed of the mass being u and

$$m(du/dt) = \mu F(t) \quad (45)$$

where F(t) is the dynamic normal contact force and can be approximated by a sine curve, i.e.,

$$F(t) = F_o\sin(\pi t/\tau)$$

and F_o and τ can be calculated from Eqs.(9) and (6) respectively.

$$mu = \mu\int_0^t F(t)dt = \mu\int_0^t F_o\sin(\pi t/\tau)dt$$

at $t = t_1$, $u = U$, i.e., **lock-up** or **no slip**

$$u = \mu F_o t[1 - \cos(\pi t/\tau)]/\pi m \quad \text{for } t < t_n < \tau \quad (46)$$

Rearranging Eq.(46),

$$\pi t/\tau = \cos^{-1}[1 - (\pi mu/\mu F_o t)]$$

and $t_1 = \cos^{-1}[1 - (\pi mU/\mu F_o t)]/\pi$ (47)

letting $f_s = \pi mU/\mu F_o t$ (slip factor)

then for $0 < f_s < 2$ slipping gives way to adherence at t_1

and for $f_s > 2$, t_1 is imaginary, i.e., slipping continues

The Zero Wear Theory states that :

$$N(1 + \beta)(\sigma_0/\omega\sigma_y)^9 = 2000$$

where N is the number of loading cycles.
β is the surface damage contribution factor due to surface and in-depth shear stresses = β_2/β_1.
σ_0 is the maximum pressure during contact.
σ_y is the uniaxial tensile yield stress,
= 1.75 s_y (hardness or shear yield stress)
ω is the wear factor, a characteristic parameter for the materials, estimated from test results.

From Hertz's contact theory

$s_{max} = 0.31\ \sigma_{max}$ at $z = 0.47$

Damage $D = K[s(t)]^n dt$

Damage contributed by a single cycle of in-depth stress

$$D_1 = K\int_0^{\tau} [0.31\sigma(t)]^n dt$$

since $\sigma(t) = \sigma_0 \sin(\pi t/\tau)$

$$\therefore \quad D_1 = K(\tau/\pi)(0.31\sigma_0)^n \int_0^{\pi} \sin^n\xi\ d\xi$$

Surface damage

$$D_2 = K\int_0^{t_1} [\mu\sigma(t)]^n dt \qquad \text{for} \quad 0 < f < 2$$

$$= K(\tau/\pi)(\mu\sigma_0)^n \int_0^{\cos^{-1}(1-f)} \sin^n\xi \, d\xi$$

$$\beta = D_2/D_1 = (\mu/0.31)^n \int_0^{\cos^{-1}(1-f)} \sin^n\xi \, d\xi \; / \int_0^{\pi} \sin^n\xi d\xi$$

For $f > 2$, β is increased in proportion to the length of sliding during τ, e.g., for $f = 2$, the lower border line case, a multiple equals to 1, i.e., $f/2$ is used.

Thus

$$D_2 = K(f/2)(\tau/\pi)(\mu\sigma_0)^n \int_0^{\cos^{-1}(-1)} \sin^n\xi \, d\xi$$

and $\beta = (f/2)(\mu/0.31)^n$

For compound impact wear, ω is given as approximately equal to 1.1 and $n = 9$ for steel on steel.

4.0 TEST RIG

The impact-fretting rig is the first apparatus designed in the Tribology and Mechanics Laboratories with the aid of the new Computer Aided Design System.

Figure 1 shows a general layout of the test rig. The essential components consist of two 300 N electro-magnetic shakers driving either one or both specimens housed inside an autoclave. With both shakers connected to one specimen, and by controlling the shakers individually or as a unit through a phase-lock frequency generator, the moving specimen can be made to impact on the stationary specimen at any desired angles. These include pure normal impact and pure reciprocating sliding.

Alternatively, with the two shakers connected individually to the two specimens, other friction and wear problems that involve one moving component striking a second moving component at 90° (such as in printing and some manufacturing processes) may be simulated and studied. This alternative arrangement will be used for another study at a later stage.

4.1 Electro-magnetic shakers

The shakers can deliver a peak force of 311 N or 200 Nrms. They cover a frequency range from dc to 7500 hz. A maximum displacement

of 9.5 mm is obtainable at frequencies up to 32 hz. The shakers will be operating at 45° facing upward.

A dual frequency generator with phase-lock, and two power amplifiers are used to control the shakers. An impedance set, consisting of a force transducer and an accelerometer, is installed between the shaker diaphragm and the specimen. The signals are stored and displayed on computer.

Two alternative systems are being considered for the feedback control. Computer software may be used to provide a constant force or any one of the motion parameters. However, its response time depends on the sampling rate. If an instantaneous response is needed, a servo-control system could be added to the shaker system.

4.2 The Specimens

The moving specimen with a spherical end is mounted on a specimen holder which is fixed to two flexible push rods at 90°, as shown in Figure 2. These pushrods are connected to the shakers by way of metallic bellows mounted on the bottom half of the autoclave.

The stationary flat specimen is mounted on a tri-axial force transducer which is attached to a semi-circular bar with its ends fixed on to the upper half of the autoclave. Glass windows for viewing are installed on the two port holes where the semi-circular bar is fixed. The specimen section of the bar passes through the central opening of the moving specimen holder [Figure 3].

4.3 The Autoclave

The autoclave is divided into three sections: the upper and the lower halves, and the observation dome. All three parts are made from cast stainless steel, CF 8. The lower section, where the bellows are attached, is firmly secured on a steel stand. Heating elements are installed inside the upper section to provide a heat source when needed. The autoclave is designed to operate in air or other gaseous environments at temperatures up to 300°C. It is also designed to maintain a low vacuum.

The dome section has two large view ports which are located for optimum viewing of the contacting surfaces. The Computer Aided Design System was used to choose the angles for the view ports so that the line of sight would encounter a minimum amount of obstacles. The superiority of the CAD system was amply demonstrated during this exercise. Under some test conditions, such as in room air, the dome section may be removed during testing to provide easier access for monitoring and observation.

4.4 Foundation

The shakers, the autoclave and its supporting frame are individually mounted on vibration mounts attached to a concrete foundation measuring 1.58 m long x 0.84m wide x 0.5 m high. This arrangement helps

to prevent extraneous vibrations from transmitting back to the test section.

5.0 PRELIMINARY TEST PROGRAMS

5.1 Normal and Oblique Impacting

Two simple test rigs are used for the preliminary investigation of impacting. The first study involved impacting between a tube specimen and a flat specimen. The tube specimen, which is attached to the end of a cantilevered tube, is excited by stepper motors driving eccentric masses, [Figure 4]. Some of the results from this study have been reported in a NRC report (Ko, et al,1985). One of the findings is that, while the normal force component during oblique impacting decreases as the angle of impact is increased (from the normal), the tangential force component peaks at incident angles around 50°. Figure 5 shows an example of this relationship between the tangential force component and the incident angle. Since wear is primarily due to shear, this finding is significant in terms of oblique impact wear.

The second study involves impact between a sphere and a plane surface. The test rig shown in Figure 6, consists of a length of plexiglas tubing, an electro-magnetic coil for releasing the sphere to be dropped, and a massive base where the plane target (together with either an uni-axial force transducer or a tri-axial force transducer) is mounted. The target may be rotated along a horizontal axis so that impacting can occur at angles from 0°(normal) to 60° from the normal. During oblique impacting, the tangential force component also peaked at incident angles around 50° [Figure 7]. Other tests revealed that the impact forces are in agreement with those predicted by the Hertzian theory for elastic impact.

5.2 Surface Topography in Oblique Impact

Tests to study the shape of indentation due to oblique impact between a hard sphere and a softer target have been carried out using the same test rig shown in Figure 6. The results will be published in a separate NRC report.

These two preliminary test series not only provide some basic information on normal and oblique impacting, they also provide needed information on the characteristics of the force transducers and their mountings. It has been revealed that the impact force signals are highly sensitive to the way these force transducers are mounted and to the natural frequencies of the system.

6.0 RESEARCH PROGRAMS

6.1 Prediction of wear due to impacting and reciprocating sliding

In earlier work, it has been shown that tube fretting wear was related to the dynamic interaction, or impact force, between the tube and its support. The measured or computer-predicted resultant

forces for different types of tube motion were statistically analysed, using histograms of force level and frequency of occurrence, to obtain a force function which was correlated with the wear rate. These histograms not only characterise the type of tube motion but also reveal changes in force levels due to changes in either excitation or tube-support clearance. However, this approach does not differentiate the significance of the shear and the normal components on wear. Arbitrary weighting coefficients had to be assigned to the different levels of the resultant force in order to obtain a good wear-force correlation.

In a later approach, the x-y displacement signals of the tube motion were recorded simultaneously as those of the force signals. Thus, the tube trajectory during tube/support contact can be determined and the force signals can be resolved into normal and tangential components. The wear rate correlates well with the force function derived from the tangential component.

In practice, dynamic systems including heat exchangers are usually subjected to random motions; many parameters, that normally remain constant in an ideal test case, do vary. Therefore, a wear prediction approach that makes use only the predicted or measured dynamic interaction, such as forces, between contacting components would be a useful tool.

The programs consist of:

(1) Improvement of the tube fretting wear prediction technique. This will include investigation of the contribution of the normal component, the relationship between the contact distance and the frequency of occurence.

(2) Development of a more generalized predictive model so that it is also applicable to other mechanical components or geometrical configurations. One example of these would be the sucker rod/tube wear in oil wells.

(3) Inclusion of material properties in the predictive model. This will make the model less dependent on prior experimental wear data, and is the first step towards a design oriented wear predictive model.

(4) Linking the developed wear model to a vibration predictive model which would have been developed elsewhere. This will provide a complete package that can be used directly by designers or for failure prediction.

6.2 Laboratory simulated impact and sliding wear tests

The experimental part of the program serves two purposes: validation and exploration. Obviously, the predictive models discussed in Section 6.1 need to be validated. But to improve and to develop new models many exploratory tests are also needed so that the mechanics involved during various combinations of impacting and reciprocating sliding can be understood.

Impacting tests : velocity, force, incident angle of impact.

Reciprocating sliding tests : frequency, amplitude, normal force.

Combined impacting and reciprocating sliding tests : frequency, and amplitude of both impact and sliding motions, normal and impact forces.

Other parameters : material combinations, environmental conditions including temperature.

7.0 REFERENCES

BLEVINS, R.D., Jan.,1983
Vibration Induced Wear of Heat Exchanger Tubes
Trans. ASME, J. of Engineering Materials and Technology,
107, #1, 61-67

ENGEL, P.A., LYONS, T.H. and SIRICO, J.L., Feb.,1973
Impact Wear Model for Steel Specimens
Wear, 23, 185-201

ENGEL, P.A., July, 1974
Analysis and Design for Zero Impact Wear
Trans. ASME, Journal of Lubrication Technology, 96F, #3, 455-463

ENGEL, P.A., 1976
Impact wear of Materials
Elsevier, The Netherlands

ENGEL, P.A., June,1978
Percussive Impact Wear
Tribology International, 11, #3, 169

FRICK, T.M., SOBEK, T.E. and REAVIS, J.R., Dec.,1984
Overview on the Development and Implementation of Methodologies
to Compute Vibration and Wear of Steam Generator Tubes
Symposium on Flow-Induced Vibrations, ASME Winter
Annual Meeting, 3, 149-161

FRICKER, H.W., May, 1972
Fretting in Tube Supports of Heat Exchangers
Conf. on "Component Design in High Temperature Reactors
Using Helium as a Coolant", Inst. of Mech. Eng., London

GOLDSMITH, W., 1960
Impact
Edward Arnold, London

GOODMAN, L.E., 1959
A Review of Progress in Analysis of Interfacial Slip Damping
Structual Damping, ed., J.E. Ruzicka, ASME

HUNTER, S.C., 1957
Energy Absorbed by Elastic Waves During Impact
J. of Mechanics and Physics of Solids, 5, 162-171

HURRICKS, P.L., 1970
The Mechanism of Fretting - A Review
Wear, 15, 389-409

JOHNSON, K.L., 1963-1964
Mechanics of Fretting
Proc. I.Mech.E., London, 178, pt 3F, 7

JOHNSON, K.L., 1985
Contact Mechanics
Cambridge University Press, Cambridge

KO, P.L., TROMP, J.H. and WECKWERTH, M.K., 1982
Heat Exchanger Tube Fretting Wear: Correlation of Tube Motion and Wear
ASTM Special Technical Publication 780 "Materials Evaluation Under Fretting Conditions", 86-105

KO, P.L. & BASISTA, H., Feb.,1984
Correlation of Support Impact Force and Fretting Wear for a Heat Exchanger Tube
Trans. ASME, J. of Pressure Vessel Technology,106, #1, 69-77

KO, P.L., Nov.,1985
The significance of Shear and Normal Force Components on Tube Wear due to Fretting and Periodic Impacting
Wear, 106, #1-3, 261-281

KO, P.L. and BOWMAN, I., Oct.,1985
The Significance of Normal and Shear Force Components on Tube Wear due to Oblique Impacting
NRC Report TR-WE-44, NRC No. 25256

LEVY, G. and MORRI, J., Nov.,1985
Impact Wear in CO_2 -based Environments
Wear, 106, #1-3, 97-138

MAW. N., June,1976
The Oblique Impact of Elastic Spheres
Wear, 38, #1, 101-114

MINDLIN, R.D., Sept.,1949
Compliance of Elastic Bodies in Contact
J. of Applied Mechanics, Trans.A.S.M.E., 71, pp 259-268

RABINOWICZ, E. and Hozaki, K., July,1985
Impact Wear of Ductile Metals
Proc. of the JSLE International Tribology Conference, 2, 263-268

RICE, S.L., June,1979
The Role of Microstructure in the Impact Wear of Two Aluminum Alloys
Wear, 54, #2, 291-301

RICE, S.L., Nowotny, H. and Wayne, S.F.
Formation of Subsurface Zones in Impact Wear
ASLE preprint No. 80-AM-2D-3

RICE, S.L., Wayne, S.F. and Nowotny, H., 1980
Material Transport Phenomena in the Impact Wear of Titanium Alloys
Wear, 65, #2, 215-226

SPROLES, E.S. Jr., GAUL, D.J. and DUQUETTE, D.J., 1978
A New Interpretation of the Mechanism of Fretting and Fretting Corrosion Damage
Proc. Int. Conf. on the Fundamentals of Tribology, MIT Cambridge, MA, 585-596

STOWERS, I.F. and RABINOWICZ, E., 1973
The Mechanism of Fretting Wear
Journal of Lubr. Technology, Trans.ASME, 95, 65-70

SUH, N.P., July,1973
The Delamination Theory of Wear
Wear, 25, 111-124.

TIMOSHENKO, S. and GOODIER, J.N., 1969
Theory of Elasticity, 3rd edition, McGraw-Hill

WATERHOUSE, R.B., 1972
Fretting Corrosion
Pergamon Press

WATERHOUSE, R.B., Dec.,1977
The Role of Adhesion and Delamination in the Fretting Wear of Metallic Materials
Wear, 45, #3, 355-364

WELLINGER, K. and BRECKEL, H., 1969
Characteristic Quantities and Wear in the Impact of Metallic Materials
Wear, 13, 257-281

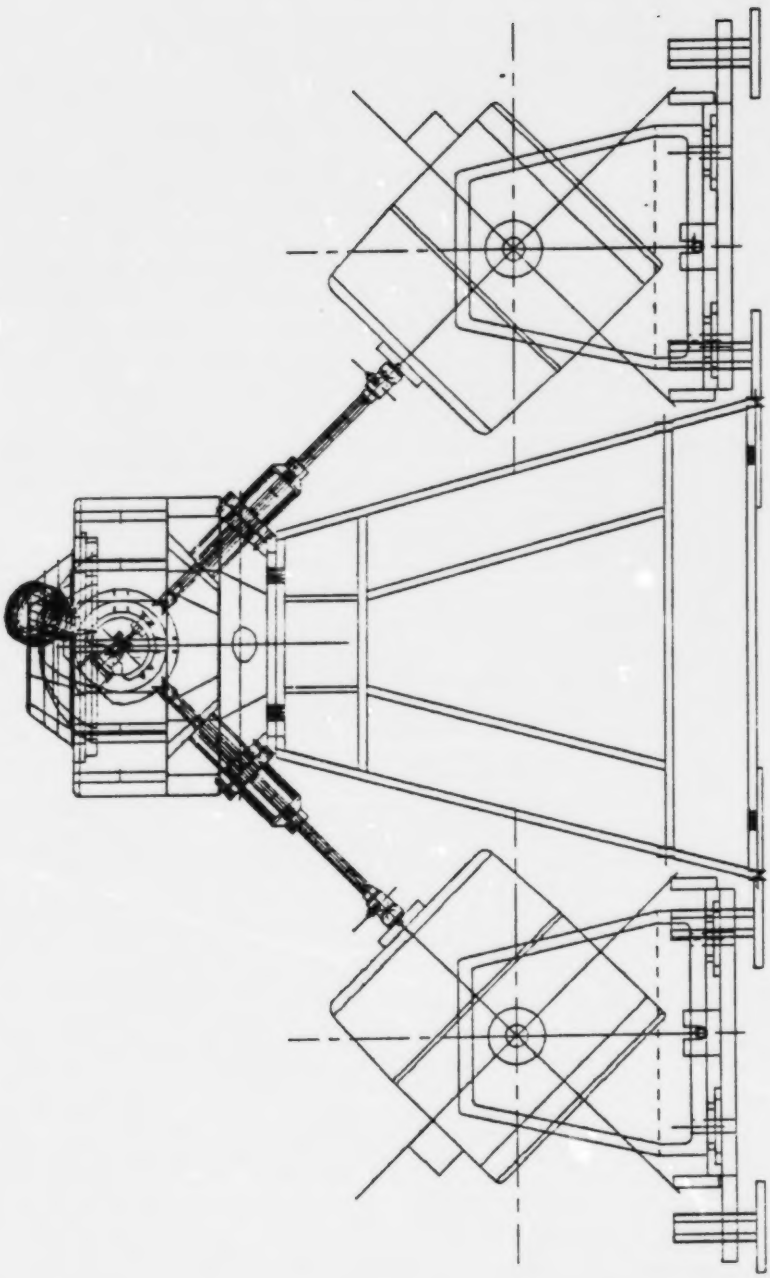


Fig. 1. General layout of impact-fretting test rig

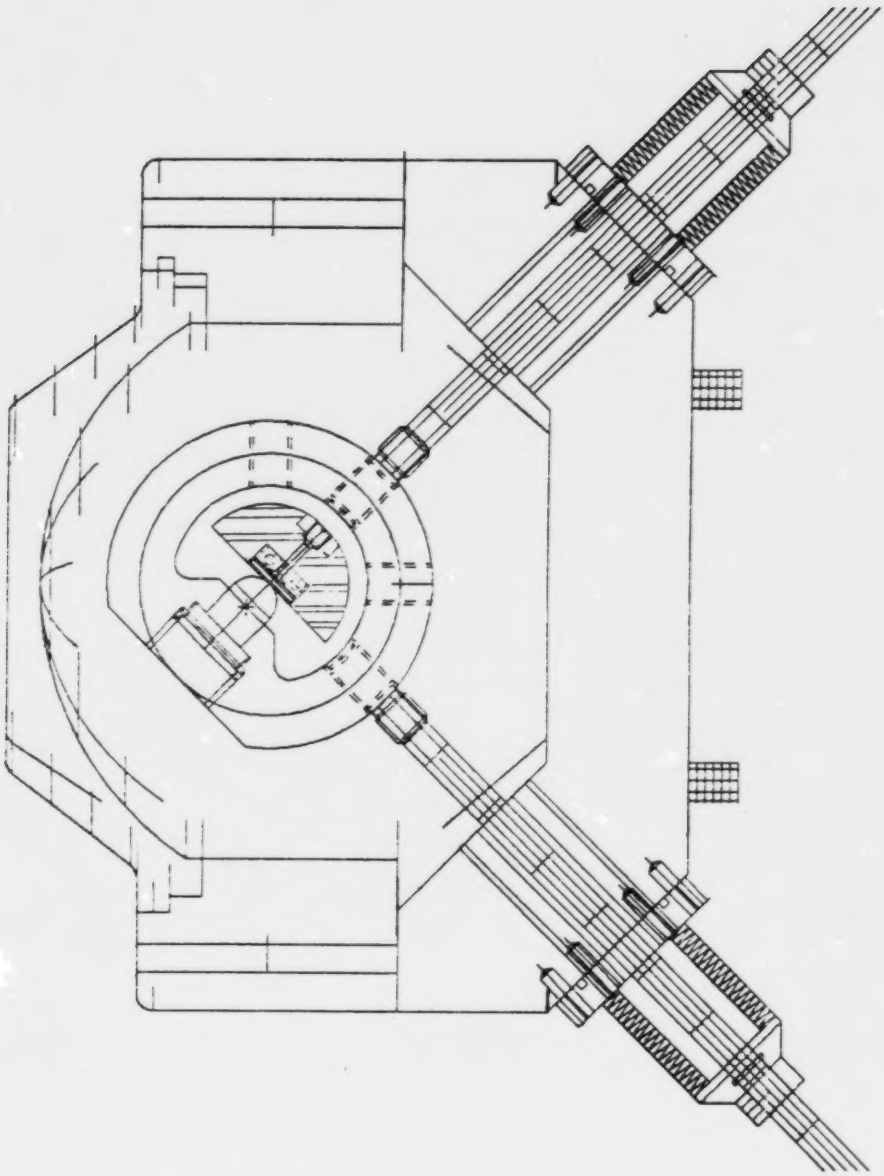


Fig. 2. Specimen-exciter connection

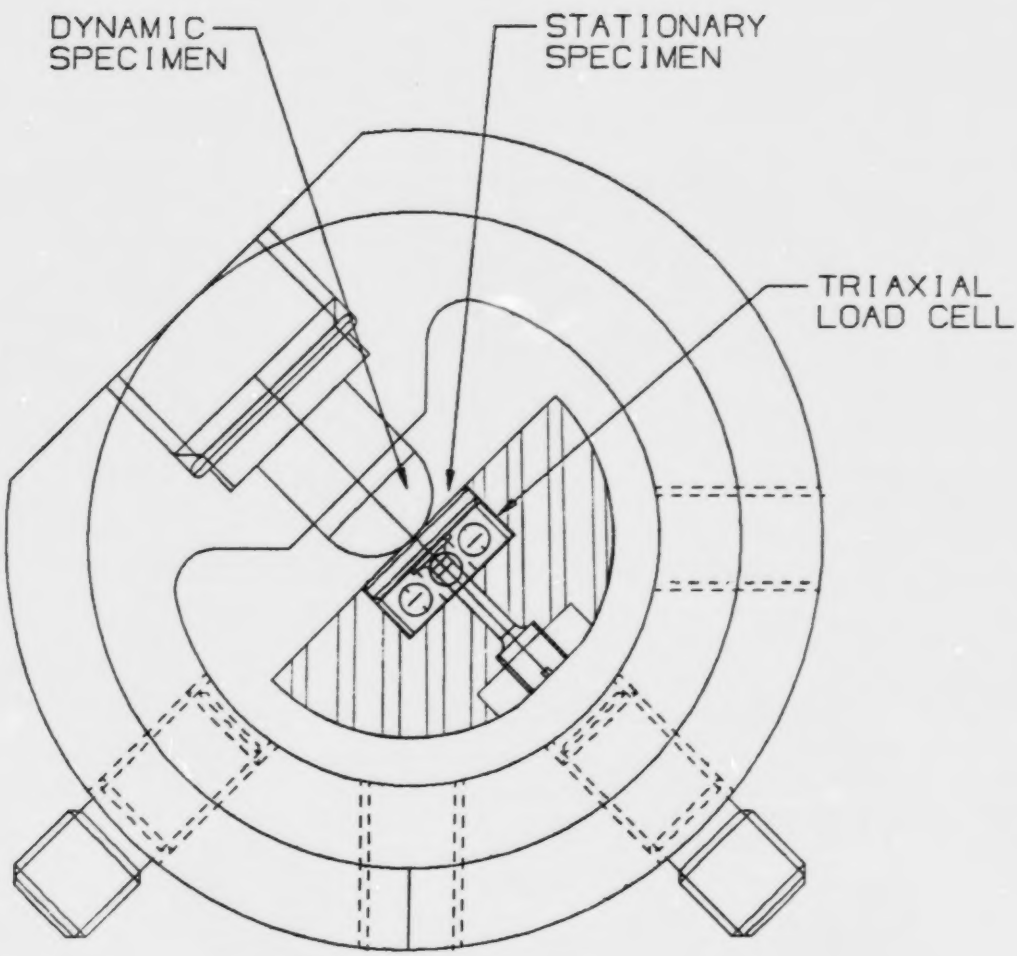


Fig. 3. Close-up view of the specimen assembly

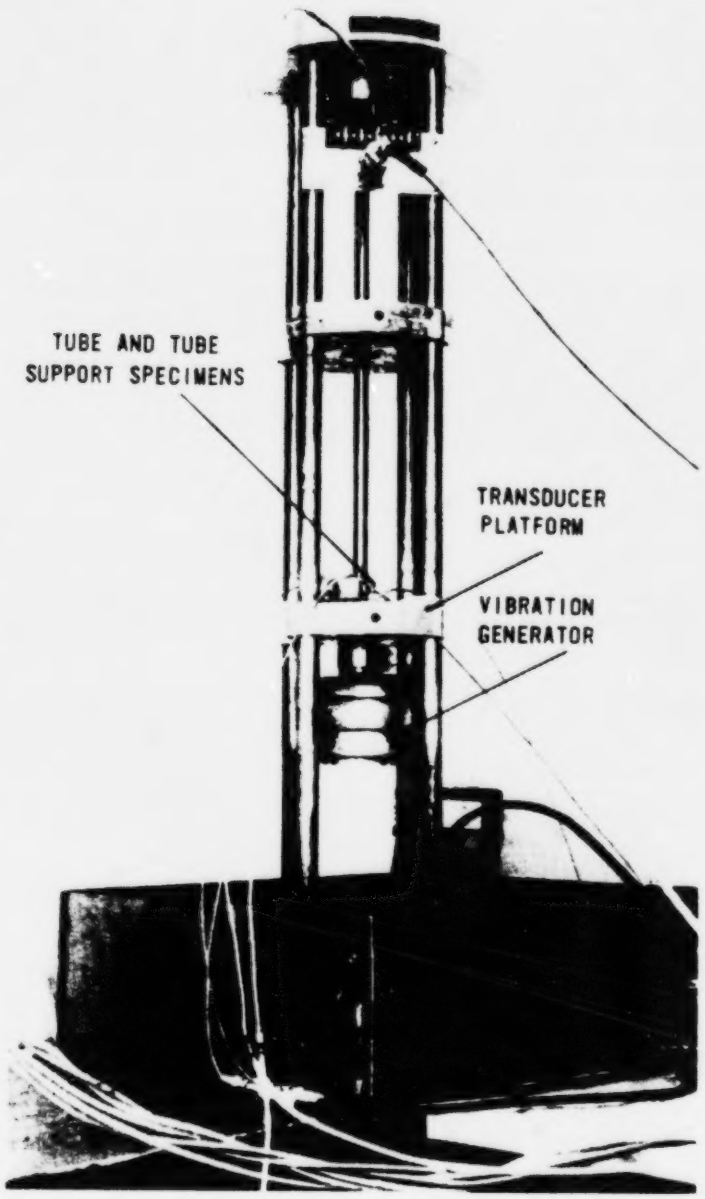


Fig. 4. Tube fretting test rig

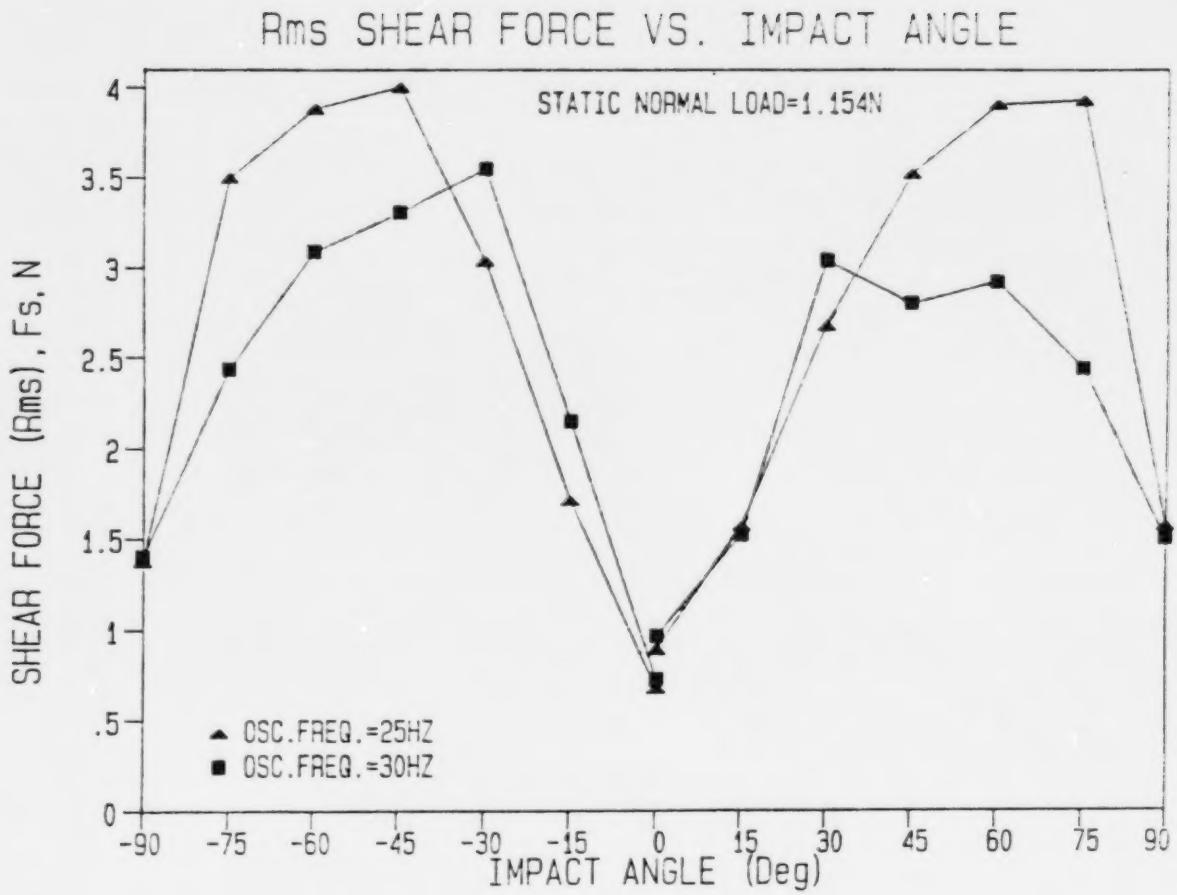


Fig. 5. Tangential forces vs impact angle from restrained tube impact tests

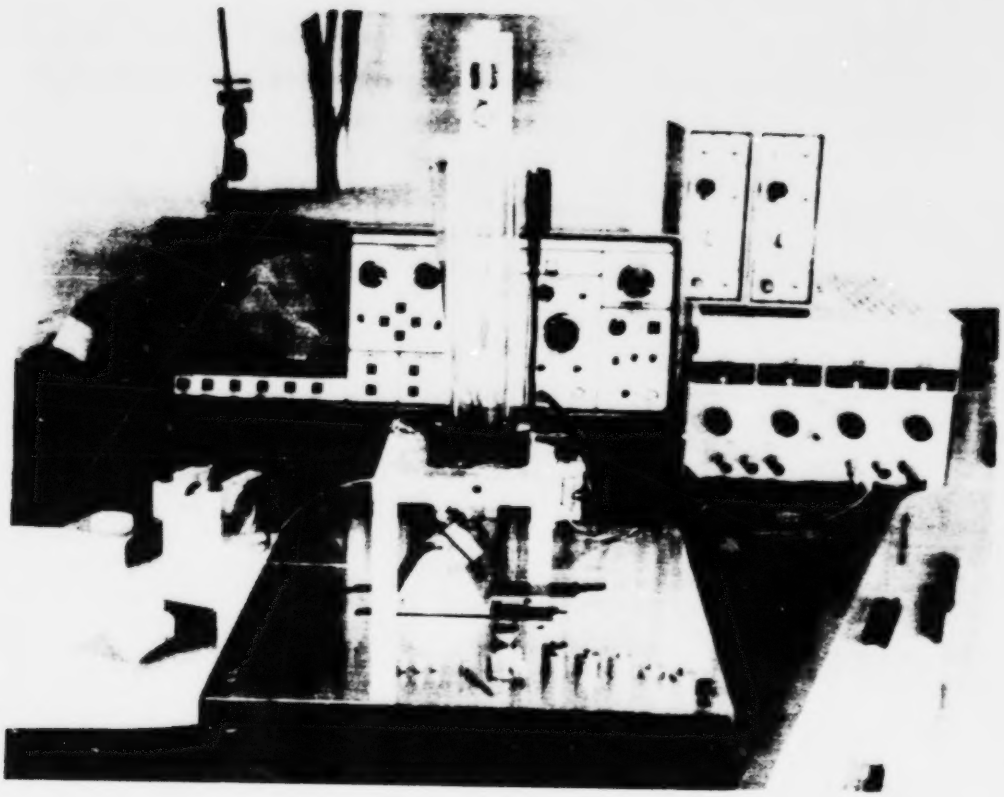


Fig. 6. Oblique impact test rig

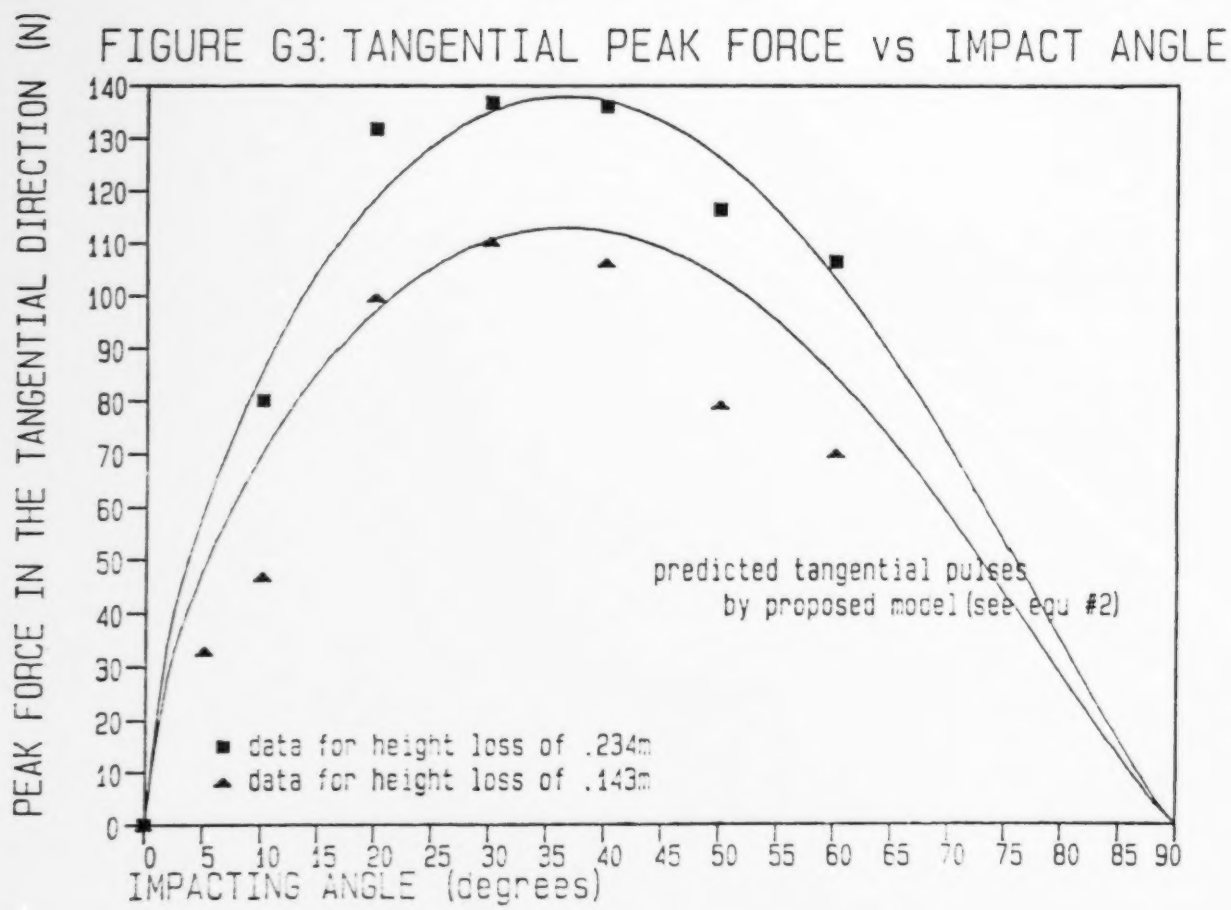


Fig. 7. Tangential forces vs impact angle from unrestrained oblique impact tests

REPORT DOCUMENTATION PAGE/PAGE DE DOCUMENTATION DE RAPPORT

REPORT/RAPPORT TR-TM-004 1a	REPORT/RAPPORT NRC No. 27948 1b
SECURITY CLASSIFICATION/ CLASSIFICATION DE SÉCURITÉ Unclassified 2	DISTRIBUTION/DIFFUSION ☐ Controlled/Contrôlé ☒ Unlimited/Illimité 3

TITLE/SUBTITLE/TITRE/SOUS-TITRE
IMPACT AND FRETTING WEAR STUDIES
Part 1 - TEST RIG AND REVIEW OF THEORIES
4

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6

CORPORATE AUTHOR/PERFORMING AGENCY/AUTEUR D'ENTREPRISE/AGENCE D'EXÉCUTION
National Research Council
Division of Mechanical Engineering
Tribology and Mechanics Laboratory
7

SPONSORING AGENCY/AGENCE DE SUBVENTION
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DATE	FILE/DOSSIER	LAB. ORDER COMMANDE DU LAB.	PAGES	DIAGS	REFS
1987/04			29	7	31
9	10	11	12a	12b	12c

NOTES
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DESCRIPTORS/KEY WORDS/MOTS-CLÉS
Impact, Wear, Fretting, Test Rig, Tribology
14

SUMMARY/SOMMAIRE
Describes a new test rig for impact-fretting studies. Reviews basic theories involved in impact and fretting wear predictions.
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